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Measurement of the pressure variations in the vapor passage of a simulated heat pipe

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MEASUREMENT OF THE PRESSURE VARIATIONS IN THE VAPOR PASSAGE OF A SIMULATED HEAT PIPE

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This work is dedicated to

Prof. Edward Kenneth Levy

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I would like to express my gratitude to Dr. Edward Kenneth Levy, first, as an inspiring teacher, and second, as my adviser who has given trenchant advice and comments throughout the course of the thesis program. It has indeed been a privilege to discuss the experimental work and results with him.

I am indebted to my wife, Madhu, whose patience and encouragement made this work such a pleasant undertaking.

I would like to thank Mrs. Beverly Spalding for her careful attention in typing this thesis.

To all these people I am deeply grateful.

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NOMENCLATURE

d	=	diameter
h	=	width of two-dimensional channel
L	=	half length of the heat pipe
M	=	momentum
p	=	dimensional pressure
P	=	dimensionless pressure
R	=	$\frac{\rho h v_w}{\mu}$
r	=	radius
u	=	dimensional axial velocity
$\bar{u}_{max}$	=	dimensional velocity at $x = L$
$\bar{u}$	=	average axial velocity
v_w	=	transverse velocity at $y = h$
x	=	axial coordinate
y	=	transverse coordinate

Greek letters

ρ	=	density
μ	=	dynamic viscosity
ν	=	kinematic viscosity
τ_w	=	wall shear stress

ABSTRACT

An experimental and analytical study of the axial pressure variations in the vapor passage of a simulated annular heat pipe is described. The outer wall of the simulated heat pipe is solid and the mass suction and injection occur at the inner porous wall; the radius ratio ($r_i/r_w = 0.87$) is close enough to unity to approximate the flow in plane two-dimensional heat pipes.

A two-dimensional channel flow control volume analysis assuming a parabolic velocity profile is developed to predict the axial pressure distributions. The axial pressure distributions predicted by the analysis show good agreement with the experimentally measured axial pressure variations in the range $0 \leq R \leq 2.5$ (R = wall Reynolds number). For the values of $R > 2.5$, good agreement between the theoretically predicted pressure distributions and the experimentally measured pressure variations occurs only in the evaporator zone of the simulated heat pipe.

It is also observed that for $R > 2.5$ the present analysis overestimates both the axial pressure recovery in the condenser zone and the overall pressure drop in the heat pipe.

CHAPTER I

INTRODUCTION

Heat pipes are structures of very high thermal conductance. The heat is transferred as latent energy by the evaporation of the working fluid in a heating zone (evaporator) and the condensation of the vapor in the cooling zone (condenser) (see Figure 1).

The flows in the vapor passage and wick region are accompanied by pressure drops which must be supported by the capillary pumping capacity of the wick structure if the heat pipe is to operate properly. Therefore, a knowledge of the overall pressure drop in the vapor passage is needed in designing a heat pipe.

In addition to the cylindrical configuration described in the preceding paragraphs, heat pipes are also built with vapor passages that have other shapes. For a complete description of the various heat pipe geometries and of the operating principles of the heat pipe see Reference [1].

The flow in the vapor passage of the heat pipe can be modeled by the flow of a fluid in a porous duct with mass transfer at the wall. In order to predict the pressure variations in heat pipes, attempts have been made in the past to solve the Navier Stokes equations and the boundary layer equations for various geometries.

The similarity solutions for the evaporator and the condenser regions were developed by Berman [2,3], Yuan [4] and White et al [5]. For the case of a round tube, entry region solutions have been obtained by Busse [9], Bankston and Smith [10], and Quaile [6].

The pressure drop equations used in heat pipe design calculations are for the most part obtained from the similarity analyses. These include for example equations by Knight [8].

There are a number of studies dealing with experimental pressure measurements for the case of either suction or injection. These include investigations by Bundy and Weissberg [11], Quaile [6,7], and Weissberg et al [12].

For flow through round ducts, closed at the ends with injection occurring over half the duct length and suction over the other half, Bankston and Smith [10] demonstrated that for $0 < R < 10$ the pressure drop obtained from the finite difference solution of the Navier Stokes equations is in good agreement with the pressure drop calculated assuming a parabolic velocity profile everywhere in the duct. Similar theoretical results were obtained by

Busse [9] for the circular geometry. Up to the present time, there has been no experimental verification of the validity of this simplified technique for estimating overall pressure drop.

The objective of this study is to conduct such an investigation for an annular heat pipe geometry. Experimental data on axial pressure profiles and overall axial pressure variations are obtained and show good agreement with simplified two-dimensional channel flow theory in the range $0 \leq R \leq 2.5$. For $R > 2.5$ the simplified theory underestimates the magnitude of the overall pressure drop.

CHAPTER II

ANALYSIS

Consider the steady, incompressible, laminar, two-dimensional fully developed flow through a channel of width h and length 2L. The channel is closed by impermeable walls at $x = 0$ and $x = 2L$. The fluid is injected from the upper wall with uniform velocity v_w into the channel between the planes $0 \leq x \leq L$. The fluid is removed from the channel between the planes $L \leq x \leq 2L$ with uniform velocity v_w' as shown in Figure 2.

Consider a control volume analysis to predict the pressure variations in this simulated heat pipe geometry. The balance of forces requires:

NET EFFLUX OF MOMENTUM = $\sum$ FORCES

$$M(x) - M(0) = -h[p(x) - p(0)] - \left[\int_0^x \tau_w dx\right]_{y=0} - \left[\int_0^x \tau_w dx\right]_{y=h} \qquad (2.01)$$

or

$$p(x) - p(0) = -\rho\overline{u}^2(x) - \left[\int_0^x \frac{\tau_w}{h} dx\right]_{y=0} - \left[\int_0^x \frac{\tau_w}{h} dx\right]_{y=h}$$

or

$$\frac{p(x) - p(0)}{\frac{1}{2}\rho \bar{u}^2_{max}} = -2\frac{\bar{u}^2(x)}{\bar{u}^2_{max}} - \frac{2}{h\rho\bar{u}^2_{max}}\left\{\left[\int_o^x \tau_w dx\right]_{y=0} + \left[\int_o^x \tau_w dx\right]_{y=h}\right\} \tag{2.02}$$

Computation of Shear Stress

The formidable task of solving Navier Stokes Equations is not undertaken here. However, as a first step it is suggested that if the shear stress term may be computed from a classical fully developed channel flow analysis, perhaps a reasonable prediction of pressure variation may be developed for the heat pipe geometry.

Two-Dimensional Channel Flow

For the case of a two-dimensional channel flow with impermeable walls the Navier Stokes Equations reduce to

$$\mu \frac{\partial^2 u}{\partial y^2} = \frac{dP}{dx} \tag{2.03}$$

with boundary conditions

$$y = 0 \quad u = 0 \quad y = h \quad u = 0 \tag{2.04}$$

Integrating equation (2.03) and satisfying the boundary conditions (2.04) yields

$$u = + \frac{h^2}{2\mu} \frac{dP}{dx} [y^2/h^2 - y/h] \tag{2.05}$$

and

$$\frac{\partial u}{\partial y} = + \frac{h^2}{2\mu} \frac{dP}{dx} [2y/h^2 - 1/h] \tag{2.06}$$

From equation (2.05)

$$\overline{u} = + \frac{1}{2\mu} \frac{dP}{dx} h^2 \int_0^h (y^2/h^2 - y/h)dy$$

or

$$\overline{u} = \frac{-h^2}{12\mu} \frac{dP}{dx} \tag{2.07}$$

From equations (2.07) and (2.05)

$$u = - 6\overline{u} [y^2/h^2 - y/h] \tag{2.08}$$

Let

$$\overline{u}(x=L) = \overline{u}_{max}$$

Then conservation of mass requires:

$$0 \le x \le L \qquad \overline{u} = \overline{u}_{max} \frac{x}{L} \tag{2.09}$$

$$L \le x \le 2L \qquad \overline{u} = \overline{u}_{max}(2 - x/L) \tag{2.10}$$

Therefore shear stress

$$0 \le x \le L \qquad \tau_w = \frac{6\mu}{h} \overline{u}_{max} \frac{x}{L} \tag{2.11}$$

$$L < x \le 2L \qquad \tau_w = \frac{6\mu}{h} \overline{u}_{max}(2 - \frac{x}{L}) \tag{2.12}$$

Pressure Variations

(a) $0 \leq x < L$

From equation (2.02)

$$\frac{p(x) - p(0)}{\frac{1}{2}\rho\,\bar{u}^2_{max}} = 2\,\frac{x^2}{L^2} + \frac{4 \times 6\mu\bar{u}_{max}}{h\rho\bar{u}^2_{max}\;hL}\,\frac{x^2}{2}$$

or

$$\frac{p(x) - p(0)}{\frac{1}{2}\rho\,\bar{u}^2_{max}} = 2(x/L)^2 + \frac{12\,\nu L}{\bar{u}_{max}\;h^2}\,(x^2/L^2) \qquad (2.13)$$

But conservation of mass requires

$$h\bar{u}_{max} = v_w L \qquad (2.14)$$

From equations (2.14) and (2.13)

$$\frac{p(x) - p(0)}{\frac{1}{2}\rho\,\bar{u}^2(0)} = 2(x/L)^2 + \frac{12}{R}\,(x/L)^2 \qquad (2.15)$$

where

$$R = \frac{v_w h}{\nu} \qquad (2.16)$$

(b) $L < x \leq 2L$

$$\frac{p(x) - p(0)}{\frac{1}{2}\rho\,\bar{u}^2_{max}} = 2(2 - x/L)^2 + \frac{12}{R} + \frac{24}{R}\int_L^x \left(\frac{2}{L} - \frac{x}{L^2}\right)dx \qquad (2.17)$$

$$= 2(2 - x/L)^2 + \frac{12}{R} + \frac{24}{R}\left[\frac{2x}{L} - \frac{x^2}{2L^2} - \left(2 - \frac{1}{2}\right)\right]$$

$$= 2(2 - x/L)^2 + \frac{24}{R}\left[\frac{1}{2} + \frac{2x}{L} - \frac{x^2}{2L^2} - \frac{3}{2}\right]$$

$$\frac{p(x) - p(0)}{\frac{1}{2}\rho \bar{u}^2_{max}} = 2(2 - \frac{x}{L})^2 + \frac{24}{R}[\frac{2x}{L} - \frac{x^2}{2L^2} - 1] \qquad (2.18)$$

and the overall pressure drop

$$\left[\frac{p(x) - p(0)}{\frac{1}{2}\rho \bar{u}^2_{max}}\right]_{x=2L} = \frac{24}{R} \qquad (2.19)$$

Theoretical Results

Solutions obtained above in the range $0 \leq x \leq L$ and $L \leq x \leq 2L$ show that the pressure variations in the evaporator and in the condenser regions are parabolic in form.

The results are presented in closed form in terms of a single parameter $(v_w h/\nu)$. The results for the pressure profiles and for the overall pressure variations are presented in Figures 5, 6, 7, 8, 9, 10. The theoretical results show that the overall non-dimensional pressure drop in a heat pipe decreases with the increase in the value of the parameter (hv_w/ν).

CHAPTER III

PRESSURE MEASUREMENTS

An experimental test section was designed to measure the pressure variations along the outer wall (Figures 3,4). To generate pressure variations which are large enough to be measured by conventional techniques, a highly viscous fluid was used. The working fluid was General Electric Silicon Fluid No. SF97 which at 77°F has a kinematic viscosity of 50 centistokes, and a specific gravity of 0.963.

Equation (2.19) shows that the dimensionless overall pressure drop in an annular porous channel can be approximated as

$$\Delta p = \frac{1}{2} \rho \bar{u}_{max} \frac{24}{R}$$

where

$$R = \frac{hv_w}{\nu}$$

and mass conservation requires

$$2\pi d_o L v_w = \frac{\pi}{4} (d_o^2 - d_i^2) \; \bar{u}(x=L)$$

Using these relations the dimensional pressure drop is written

$$\Delta p = \frac{96\,\mu^2 L^2 d_i^2 R}{h^4 (d_o + d_i)^2}$$

The pressure drop is directly proportional to μ^2, L^2, d_i^2 and R and inversely proportional to h^4 and $(d_i + d_o)^2$. It was estimated that the pressure drop using air or water is too low to be measured conveniently. The magnitude of Δp for a given R and L can be significantly increased by increasing the viscosity of the fluid and decreasing the gap h. However, there are limits on h since it is necessary that the flow be laminar. As a matter of fact, μ, L, h and ρ must be chosen so that the flow is always laminar, and $0 \leq R < 15$. It was concluded that the significant variables were the kinematic viscosity of the fluid and the gap between the outer and inner cylinders.

In this study the test apparatus was designed for $0 < R < 15$ and $L/h = 7.5$, resulting in measurable pressure variations.

The final design parameters chosen are as follows:

Length of evaporator	1.5"
Length of condenser	1.5"
Gap	.19"
Outer diameter (solid wall)	2.332"
Inner diameter (porous wall)	1.935"
Viscosity	50 centistokes at 70°F
Density	60 lb/ft^3

In the flow loop, the fluid is circulated by a Worthington 5GA gear pump driven by a 5 h.p. electric motor operating at a constant speed of 1715 r.p.m. The pumping capacity of this pump and motor is 25 G.P.M. at 125 p.s.i. The rate of fluid flow was measured by Schutte and Koerting rotameters. Three flow meters were used to cover a range from 0 to 8 G.P.M. A fourth flow meter was used to extend the range from 8 to 21 G.P.M. for higher Reynolds numbers. The manufacturer's calibration data at maximum flow rate is given below.

Rotameter	Maximum Flow Rate	Accuracy % of Full Scale
4	0.9	2%
3	2.5	1%
2	8.2	1%
1	21.5	1%

The rotameters were supplied with standard scales delineated in centimeters. The calibration of each meter was checked by actual measurements.

The test section was designed so that tests could be run on overall pressure variations and on axial pressure profiles. The porous annular channel with the solid outer wall is shown in Figure 3; the inner porous tube was made from sintered bronze filter powder and was fabricated by Sintered Metal Inc. of Boston.

The outer solid wall (Figure 3) was instrumented with pressure taps for the measurement of the axial static pressure variations. These pressure taps were connected to a Validyne Model DP15 variable reluctance differential pressure transducer. A transducer indicator model CD12 was used with the transducer. The indicator generates a d.c. voltage which is proportional to the differential pressure. The d.c. voltage was measured by using a digital voltmeter during both calibration and tests. A manifold was introduced between taps and the transducer. This made it possible to measure the axial pressure profiles as well as the overall pressure drops.

The temperature of the working fluid was measured both at rotameter and test section using chromel alumel thermocouples. The e.m.f. outputs of these thermocouples were recorded on a Honeywell two-channel potentiometric recorder. The temperatures were used to evaluate the fluid densities, viscosities and flow rates.

Experimental Procedure

The flow loop for the experimental measurement of pressure variations is shown schematically in Figure 4. For the purpose of measurement of overall pressure drop, the data recorded consisted of rotameter number, float position, temperature of rotameter, temperature of test section, voltage output of pressure transducer, pressure tap identification, and voltage reading of pressure transducer at zero flow rate (zero error).

The procedure to measure pressure profile was more complex. Particular values of rotameter, float position, temperature of rotameter, and temperature of test section were chosen; then, by use of the pressure manifold system, the pressure transducer reading and zero shift readings were recorded for a given value of x/L. The pump was then stopped and the system was allowed to cool for 15 minutes. Then the pump was again started and the system was allowed to warm up to produce exactly the same rotameter scale reading and test section temperature for a different value of x/L. The process was repeated until the pressure profile was completed for a given value of R.

CHAPTER IV

CONCLUSIONS

The measurements of overall pressure drop show good qualitative agreement with the simplified control volume analysis. The magnitude of the overall pressure drop decreases with an increase in the magnitude of the dimensionless wall Reynolds number R. The control volume analysis and the experimental measurements of the axial pressure variations also show good qualitative agreement: there is a pressure drop in the evaporator as well as in the condenser at lower values of wall Reynolds number (Figures 3,4). However, at higher values of wall Reynolds number there is a pressure recovery in the condenser zone (Figures 9, 10).

From the pressure measurements of overall pressure drop and pressure profiles in a simulated heat pipe, the following general conclusions may be drawn:

1. The net pressure at the end of the condenser is always less than the pressure at the beginning of the evaporator.

2. A two-dimensional channel flow control volume analysis assuming a parabolic velocity profile shows good agreement with the measured values of overall pressure drop in the

range $0 \leq R \leq 2.5$.

3. At higher values of wall Reynolds number ($R > 2.5$) the control volume analysis predicts a lower pressure drop than is obtained experimentally.

4. At lower values of wall Reynolds number ($R \tilde{<} 6$) (Figures 7,8) the pressure decreases in both the evaporator and condenser of the heat pipe; however, at higher values of wall Reynolds number ($R \tilde{>} 6$) the pressure increases in the condenser zone.

5. The simplified analysis is in good agreement with the pressure profile in the evaporator over the entire range of R. However, it fails to give good agreement with the pressure profiles in the condenser zone except for the low values of wall Reynolds number ($0 \leq R \leq 2.5$).

TABLE I. EXPERIMENTAL DATA

v_w ft/sec	μ centistokes	$(\Delta p)_{overall}$ non-dimensional	R	$(\Delta p)_{overall}$ p.s.i.
.021	51.65	-48.47	.64	-0.0062
.023	51.17	-40.39	.72	-0.0063
.025	48.37	-32.79	.82	-0.00585
.029	46.59	-27.38	.98	-0.0063
.052	46.59	-15.89	1.75	-0.0120
.056	44.88	-15.24	1.94	-0.0129
.060	41.25	-12.07	2.28	-0.0115
.064	41.25	-11.23	2.42	-0.01214
.080	41.25	-9.38	3.03	-0.0163
.085	40.87	-8.68	3.25	-0.0166
.135	38.63	-5.64	5.46	-0.028
.146	38.63	-5.31	5.87	-0.0305
.155	38.63	-5.11	6.24	-0.0328
.168	38.27	-4.88	6.84	-0.0366
.181	38.27	-4.79	7.39	-0.0418
.200	38.27	-4.20	8.16	-0.0444
.215	48.37	-4.15	6.94	-0.0558
.224	46.59	-3.72	7.49	-0.0541
.231	44.88	-3.49	8.06	-0.0517
.240	42.83	-3.06	8.74	-0.0472
.250	40.87	-2.64	9.54	-0.0422
.258	39.00	-2.59	10.35	-0.0442
.269	37.21	-2.57	11.31	-0.0453
.263	37.21	-2.68	11.15	-0.045
.267	36.87	-2.64	11.29	-0.046

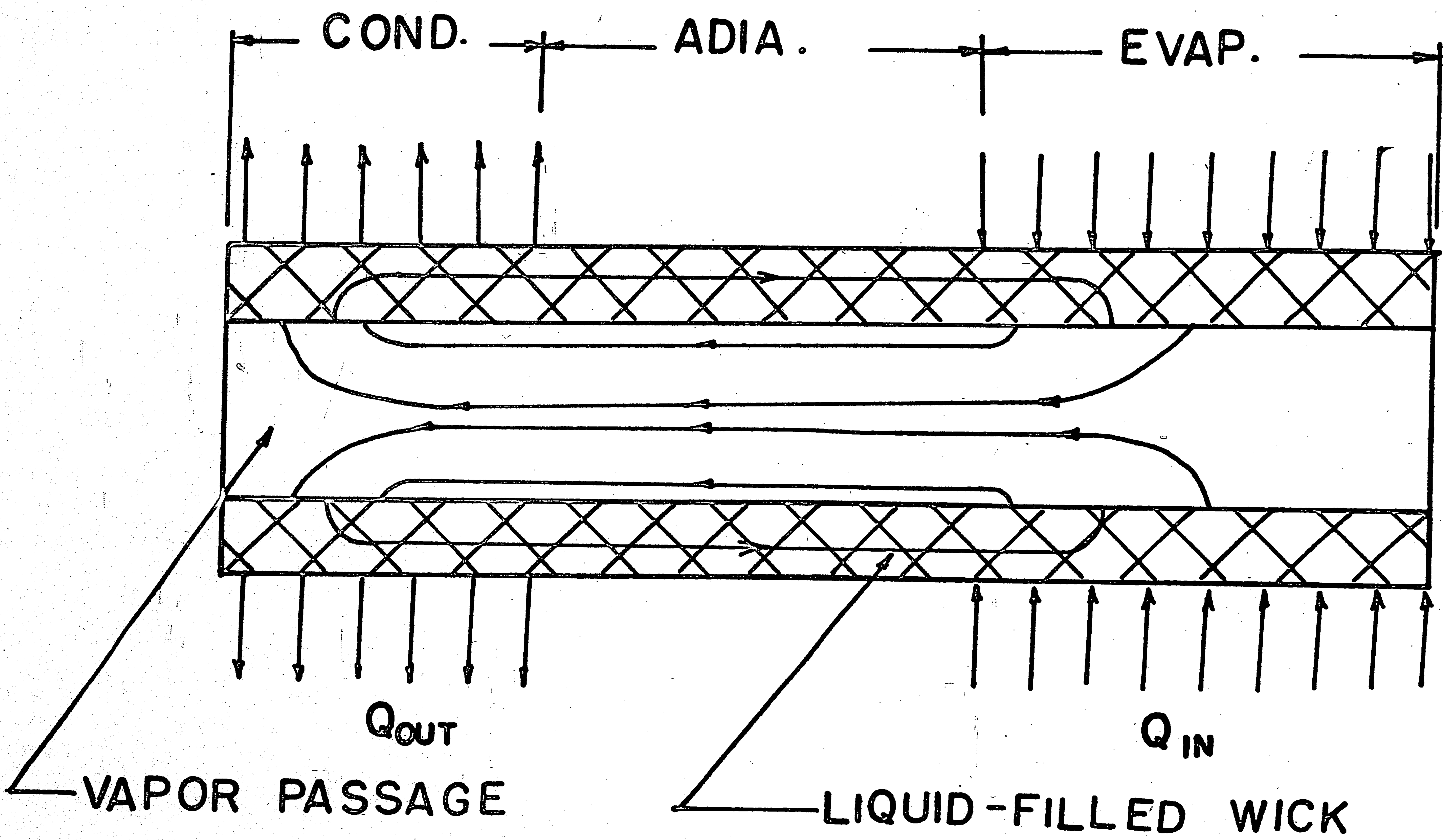

Figure 1 A Typical Heat Pipe

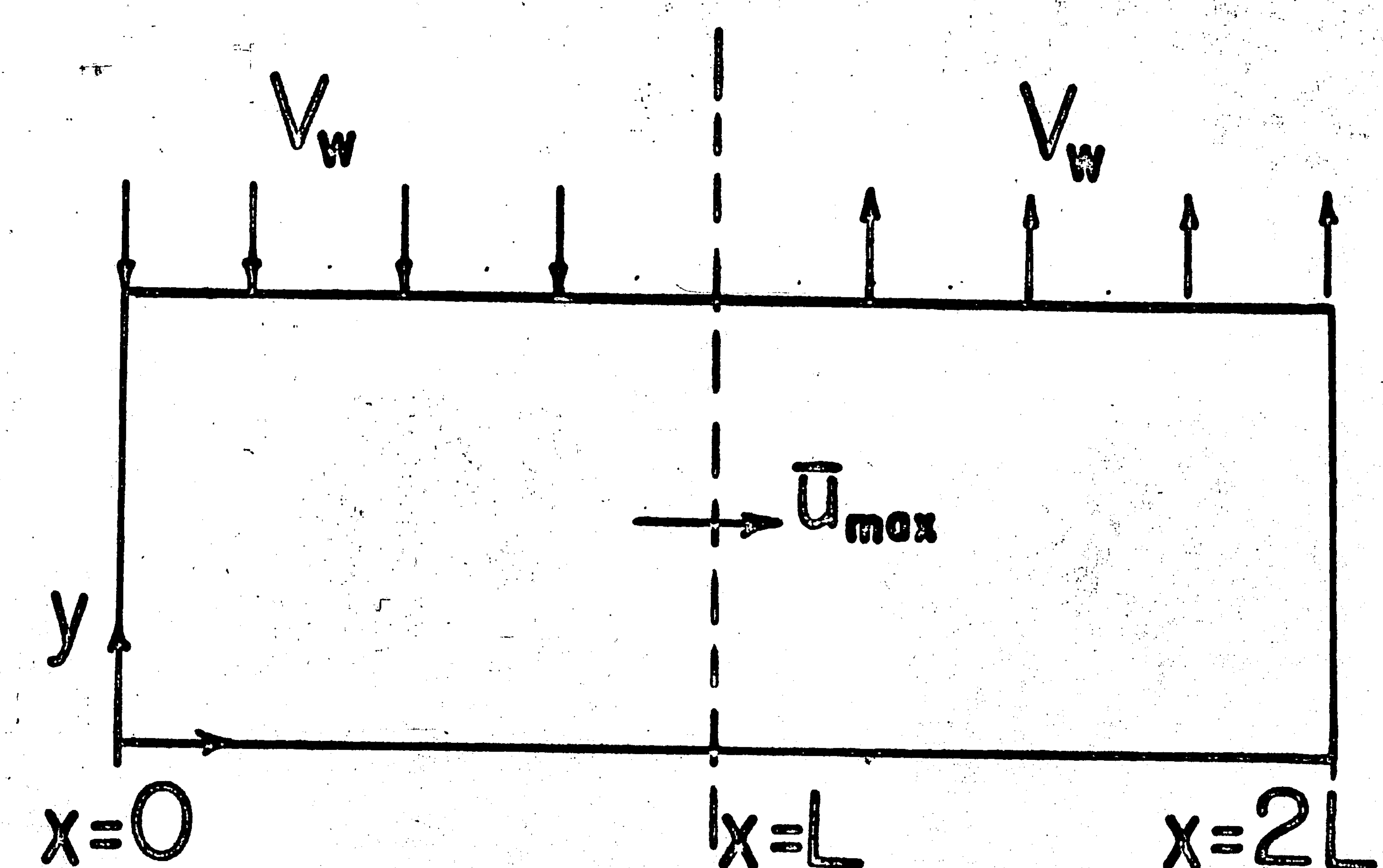

Figure 2 Coordinate System and Geometry

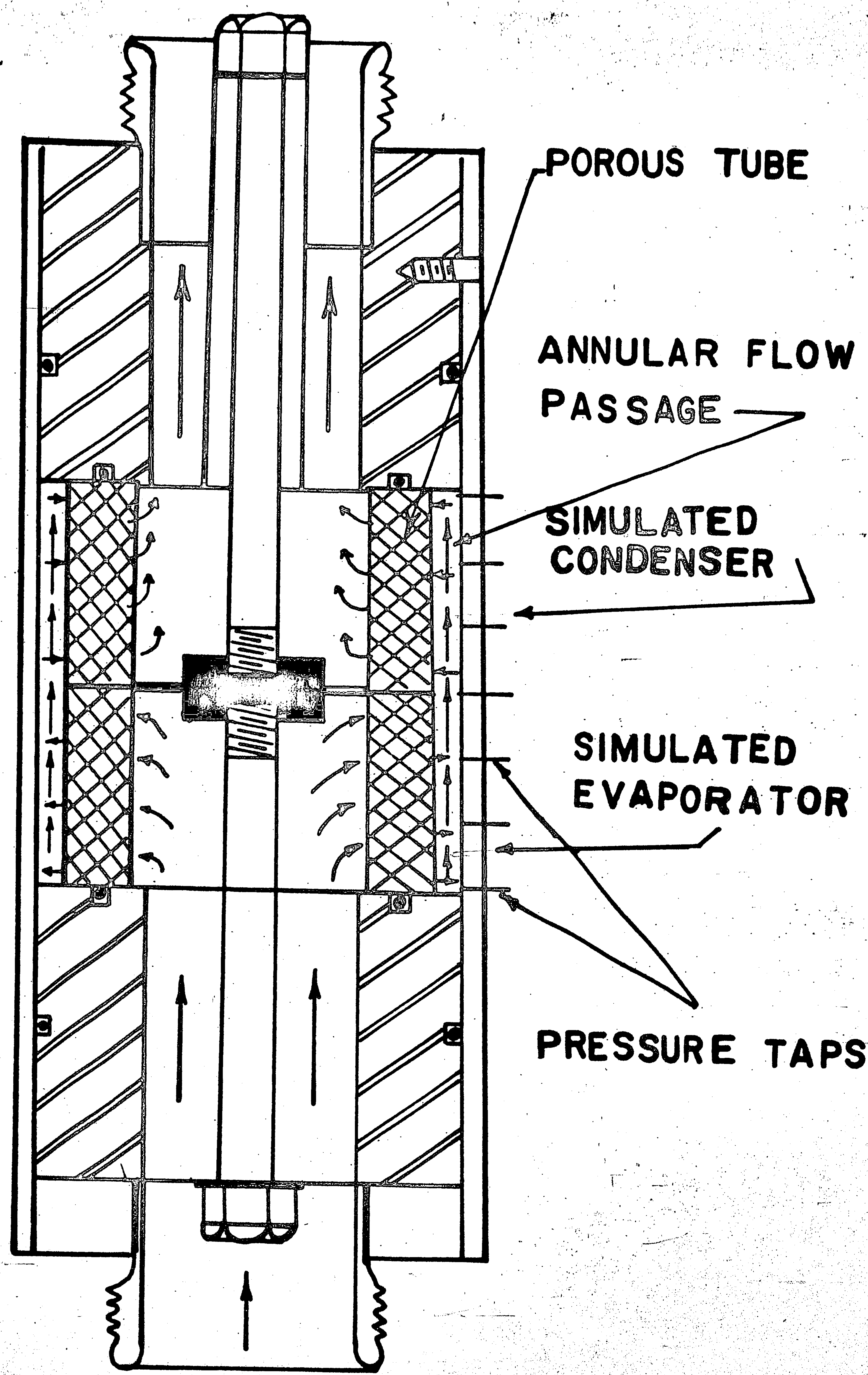

Figure 3 Test Section

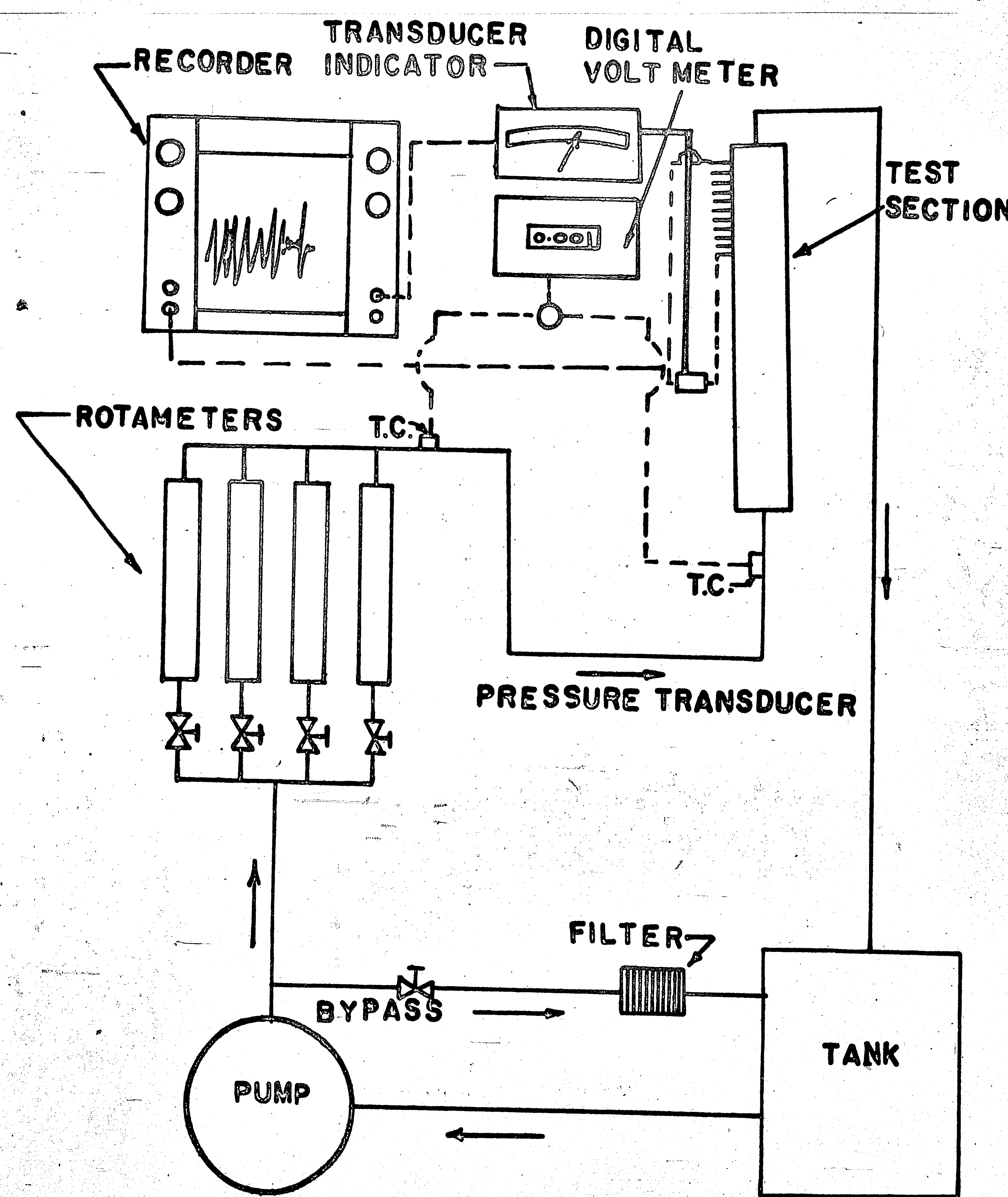

Figure 4 Test Flow Loop

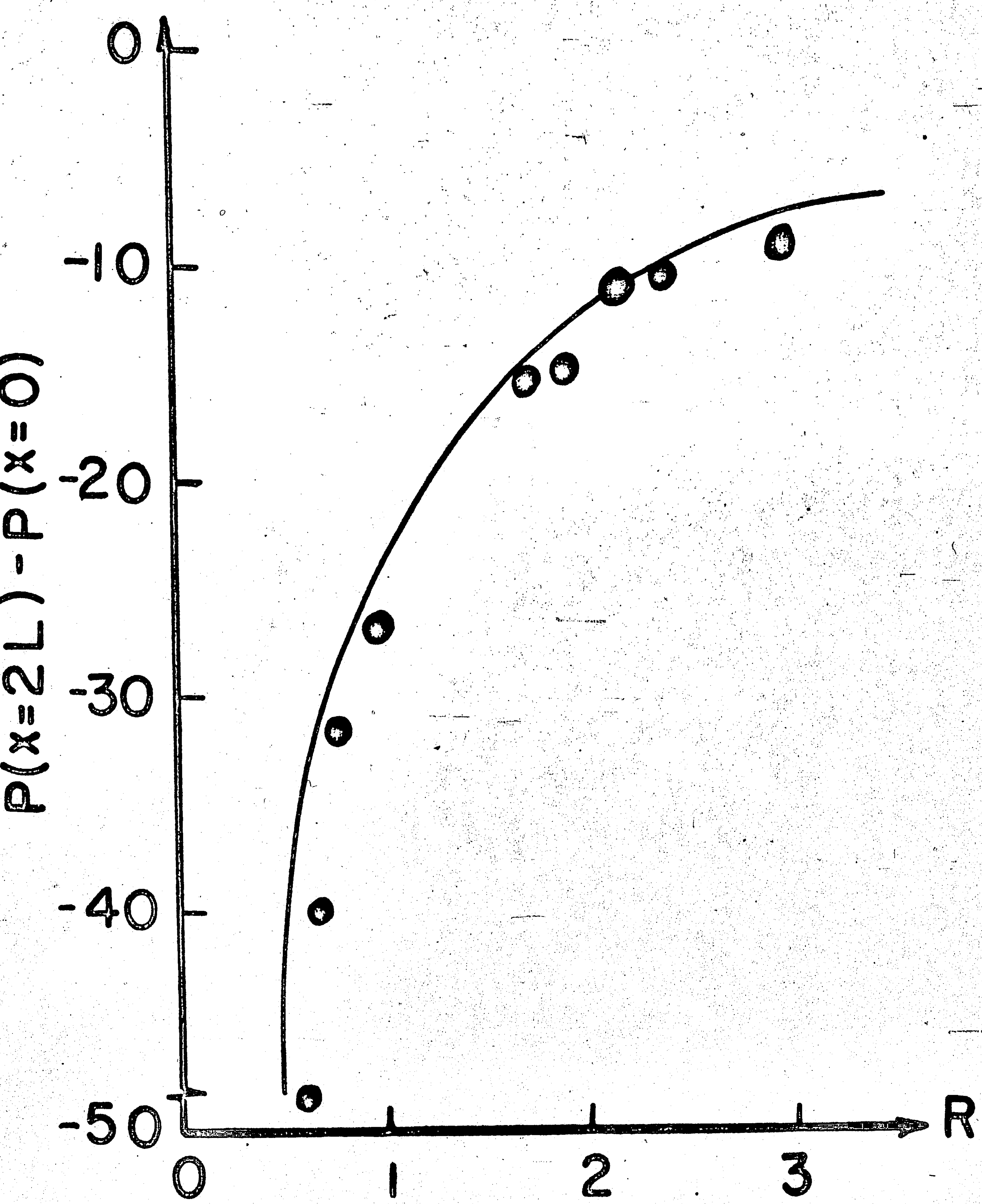

Figure 5 Overall Pressure Drop in Simulated Heat Pipe

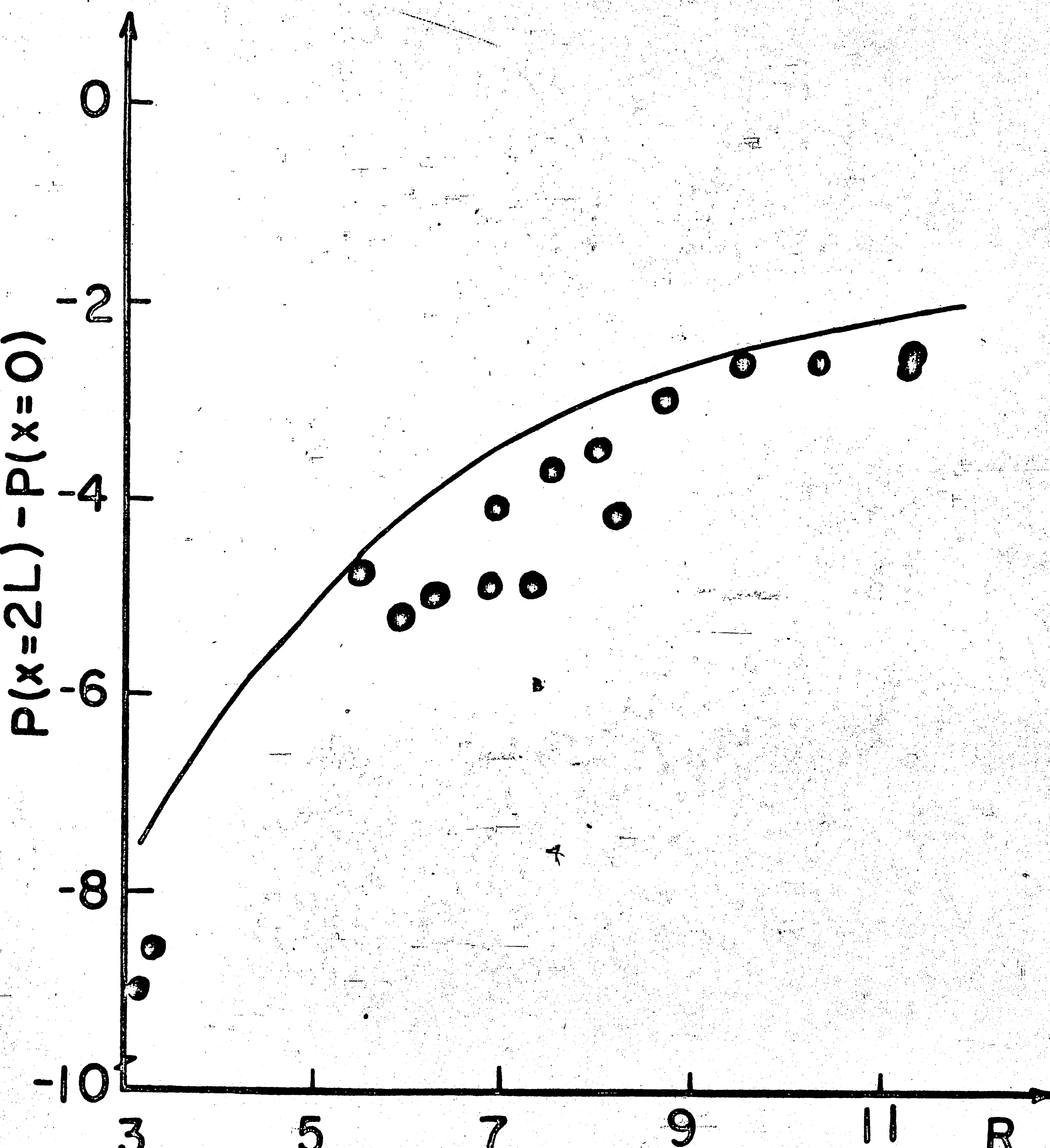

Figure 6 Overall Pressure Drop in Simulated Heat Pipe

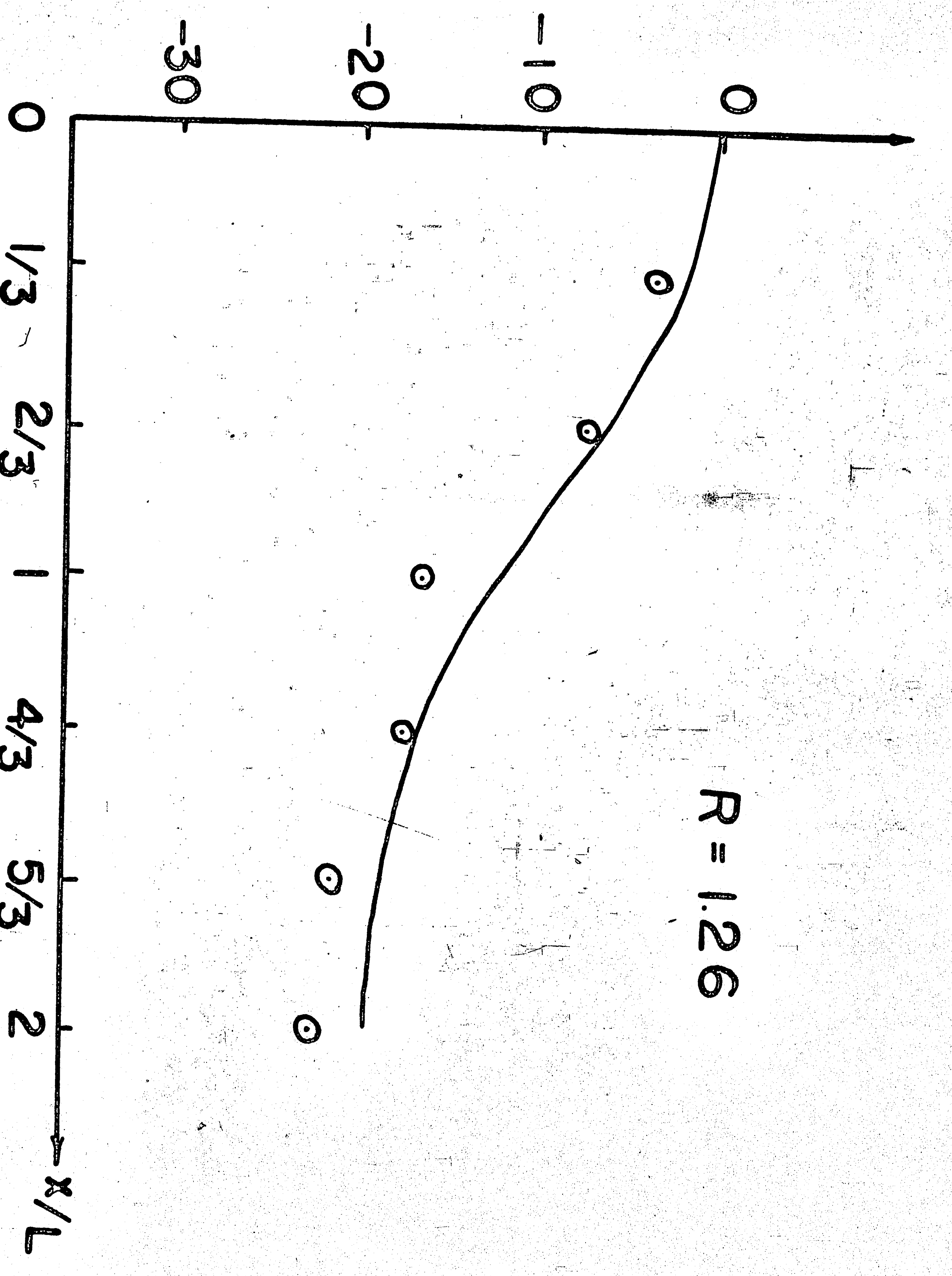

Figure 7 Axial Pressure Variation

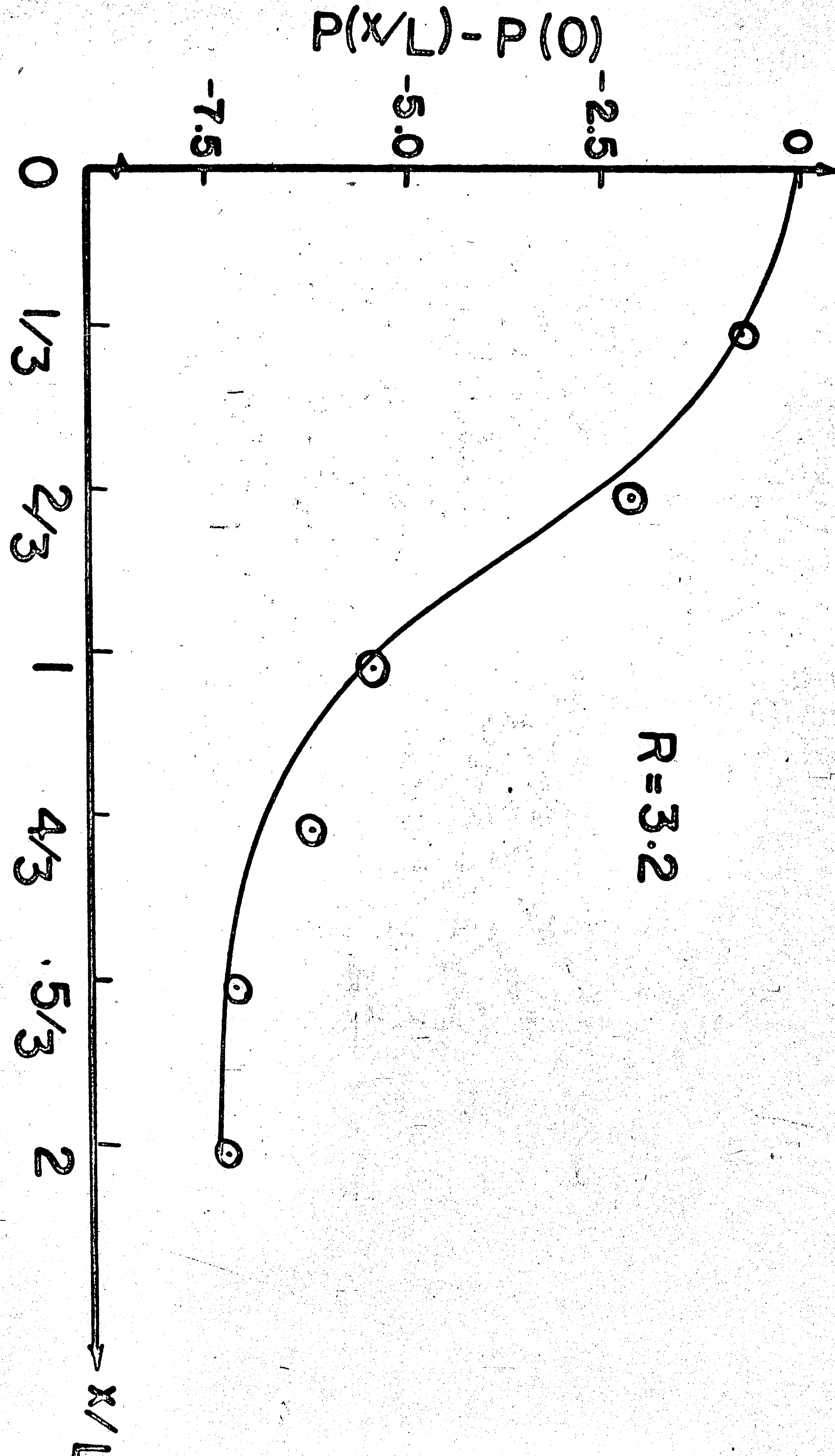

Figure 8 Axial Pressure Variation

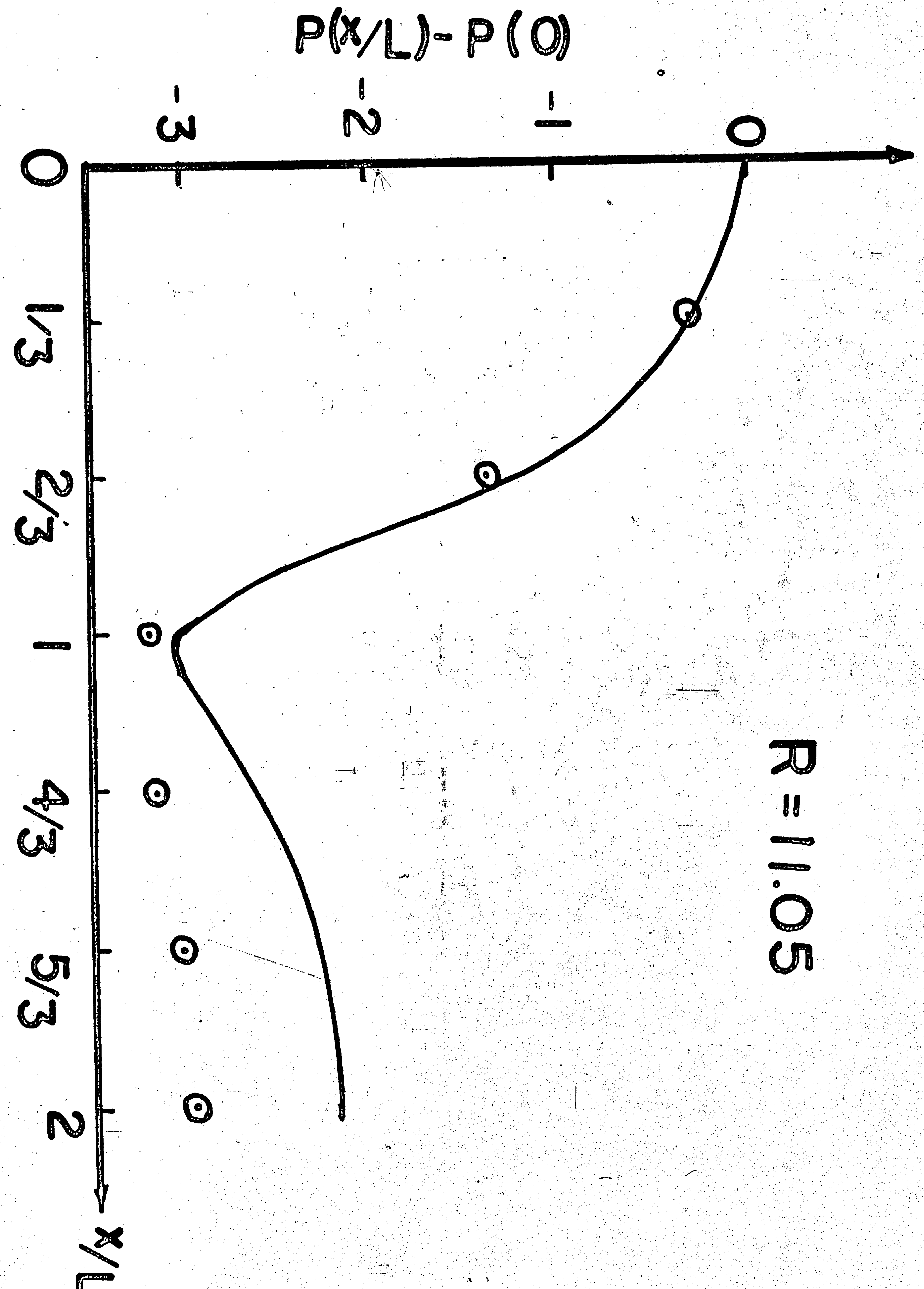

Figure 9 Axial Pressure Variation

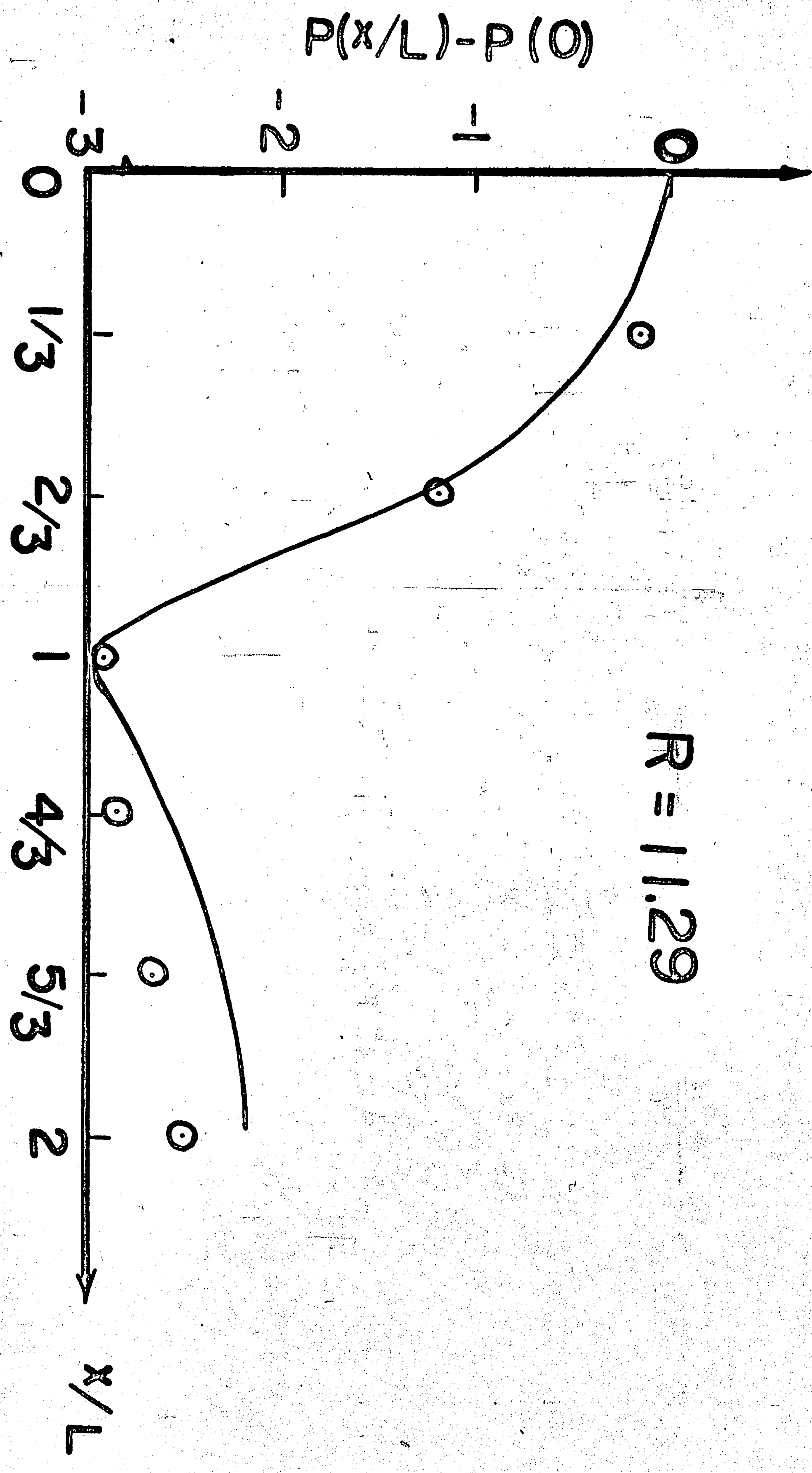

Figure 10 Axial Pressure Variation

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VITA

The author is the second son of Govind Das and Ram Kishori Gupta of Mirzapur (U.P.) India. He was born on January 1, 1943 in Mirzapur, India. After graduating from Gorkhi High School in Gwalior in 1959, the author attended Madhav Engineering College in the same city, receiving a B.Sc.Eng. (Mech.) degree in Mechanical Engineering in August 1965.

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